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JOURNAL-BEARING DESIGN AS RELATED TO MAXIMUM LOADS, SPEEDS, AND OPERATING TEMPERATURES[1]

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ABSTRACT

This paper outlines briefly a method suggested as a basis for journal-bearing design more especially for applications where the loads and speeds are variable and may reach relatively high values. This method is based upon two primary considerations; first, that the bearing shall operate in the region of stable lubrication, and second, that the operating temperature of the bearing shall not exceed some fixed value.

Equations are derived for the rate of heat generation and for the rate of heat dissipation in terms of factors of construction and operation. Equating the rates of heat generation and dissipation yields an approximate relation between the construction and operation factors and the rise in temperature of a bearing above its surroundings when operating in the region of stable lubrication. The product of the maximum allowable pressures and speeds at which a bearing will operate under the conditions prescribed for safety is obtained by substituting in the above relation permissible values for the temperature rise and the generalized operating variable ZN/P, where Z is the viscosity of the oil, N the speed of the journal, and P the bearing pressure.

In a numerical example to illustrate the application of the method, individual values for the maximum permissible pressure and speed for a given bearing when using a given lubricant are obtained from the equation for the product of the speed and pressure and the minimum permissible value of ZN/P for stable lubrication.

CONTENTS

I. INTRODUCTION

The past decade has seen a considerable advance in the science of journal-bearing lubrication. A number of mathematical and analytical treatments have extended the knowledge of the mechanics of the load-carrying oil film, and experimental investigations have justified the concept of such a film as a basis for design and have also provided correction factors for use in applying theoretical relations. The problem of bearing design, however, is complicated by the fact that a

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journal bearing not only is required to support its load under all conditions of operation, but also in many applications must provide means for dissipating the heat generated in shearing the film of lubricant.

With bearings operating at moderate loads and speeds, the heat generated in the bearing usually is small in comparison with the capacity of the bearing for heat dissipation, and the prime consideration in design is to insure that the bearing shall operate with a complete film of lubricant between the surfaces.

Rational methods [1, 2] [2] for the design of bearings operating under constant moderate loads and speeds have been described in the literature, and it is unnecessary to discuss them in detail at this time. Two methods for assuring conditions for "thick-film" or stable lubrication have been suggested. One is based upon the evaluation of the minimum thickness of the oil film by means of theoretical hydrodynamical relations involving the bearing dimensions and the assumed operating conditions. The other is based upon dimensional analysis and involves the choice of a suitable value for the generalized operating variable ZN/P, the product of the viscosity of the lubricant by the speed of the journal divided by the load per unit projected area. This variable is of particular significance in that it determines the value of both the coefficient of friction and the relative film thickness for a given bearing whenever conditions are such that the bearing is operating in the region of stable lubrication. By its use experimental data may be so correlated as to be directly applicable to design computations. It has been customary to assume a reasonable operating temperature as a basis for a first computation and by methods of successive approximations based on the relations between the heat generated, viscosity, and temperature rise, determine the operating temperature of the bearing. Recently, however, Hersey [3] has suggested a three-dimensional graphical method for the simultaneous solution of the equations involving the three variables. The final computation, of course, is accurate only to the extent that the relations between the three variables are known.

The present-day trend toward the speeding up of all types of machinery has resulted in the use of higher speeds and loads with journal bearings. In a number of applications this has reached the point where the factors affecting bearing temperature are as important as those relating to proper film formation. Many applications also involve wide variations of load and speed. In such cases the designer possibly is more interested in the extreme conditions of load and speed at which a given bearing will operate successfully than he is in its performance at normal running conditions. From this standpoint a rational basis for design would seem to be the determination of the maximum allowable loads and speeds predicated upon two primary considerations: first, that the bearing shall always operate in the region of stable lubrication, and second, that the operating temperature of the bearing shall never exceed some fixed value.

The rise in temperature accompanying the operation of a journal bearing at a given load and speed with a given lubricant decreases the viscosity, and in the region of stable lubrication will result in a decrease in the rate of heat generation. Furthermore this rise in temperature increases the rate of heat dissipation. A bearing operating in the

[2] Figures in brackets here and elsewhere in the text correspond to the numbered references at the end of this paper.

stable region therefore will tend to reach a condition of equilibrium at some temperature where the rate of heat generation is equal to the rate of heat dissipation. Hence, if the equilibrium operating temperature is to be used as a basis for design a knowledge of the factors affecting both the rate of heat generation and the rate of heat dissipation is essential.

II. HEAT GENERATED IN BEARING

Using the nomenclature given in the list of symbols, the rate of heat generation may be expressed as

$$H=k_1F\pi DN, \tag{1}$$

which may also be written as

$$H=k_1\pi fPLD^2N \tag{2}$$

Values for f, the coefficient of friction, for a given bearing at various operating conditions may be computed from theoretical hydrodynamical equations or obtained directly from experimental data. For small-bore full-journal bearings friction data obtained at the National Bureau of Standards are available. In one investigation [4] the effects of changes in the length-diameter and clearance-diameter ratios were determined over a wide range of operating conditions. The results indicated that an equation of the form

$$f=k_2(ZN/P)(D/C)+\Delta f \tag{3}$$

is reasonably accurate for the practical determination of friction losses in bearings of this type when operating in the stable region. In this equation k_2 is equal to 473×10^{-10} when the units given in the list of symbols are employed.[3] Δf is a correction for changes in the length-diameter ratio. The values to be used for various L/D ratios are shown in figure 1.

Substituting equation 3 in equation 2 yields,

$$H=k_1PND[k_2(ZN/P)(D/C)+\Delta f]\pi LD \tag{4}$$

III. HEAT DISSIPATED

In self-cooled bearings the heat is dissipated to the surroundings by a combination of radiation, convection, and conduction. Some heat also is carried away by the oil flowing through the bearing. The influence of these factors will vary with different types of bearings and different operating conditions. The available experimental data on the heat dissipation of bearings are not of such scope as to provide a rigorous evaluation of the factors involved. There is some indication [3, 5], however, that with some types of bearings the rate of heat dissipation may be approximately represented as:

$$H'=k_3A\Delta T^{1.3} \tag{5}$$

The value for the over-all coefficient of heat dissipation k_3 will depend upon many factors, including the convection currents present. Some

[3] Customary English units are employed except for Z which is expressed in centipoises following the usual practice.

data indicate that when the bearing is in a strong draft the value of k_3 may be 3 or 4 times as great as when the bearing is in still air.

Equation 5 may also be written

$$H' = k_3 k_4 \pi L D \Delta T^{1.3} \tag{6}$$

where k_4 is the ratio between the effective area for heat dissipation and the working area in the bearing. Since for equilibrium operating

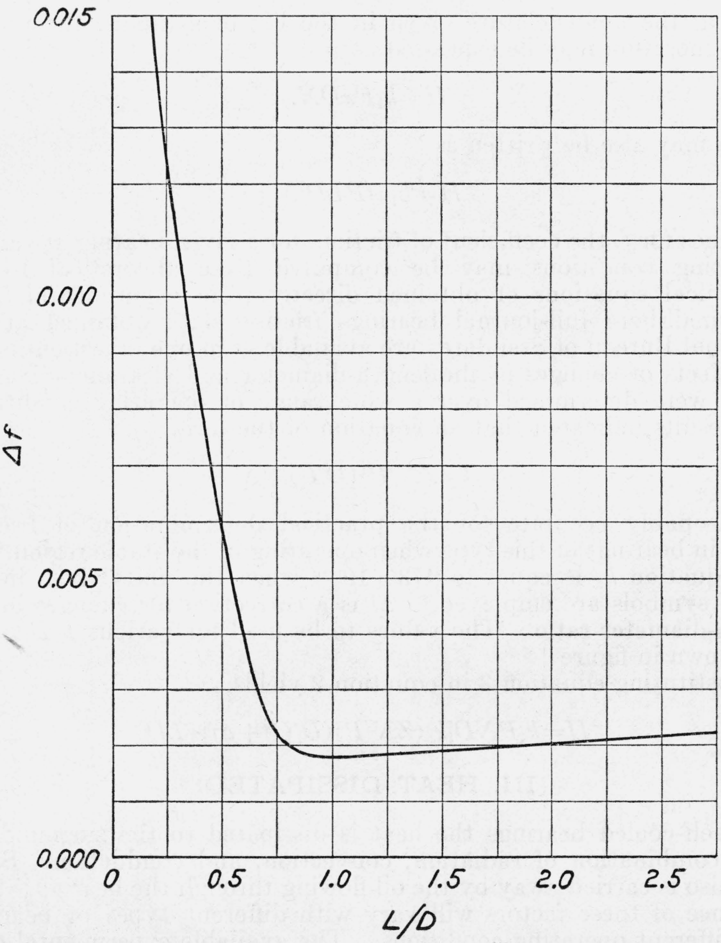

FIGURE 1.—*Values of Δf for various L/D ratios, small-bore 360° bearings.*

conditions, the rate of heat dissipation must equal the rate of heat generation, equations 4 and 6 may be combined to form

$$k_1 PND[k_2(ZN/P)(D/C) + \Delta f] = k_3 k_4 \Delta T^{1.3} \tag{7}$$

This equation indicates approximately the relation between the rise in temperature of a bearing above its surroundings and the other factors involved, when it is operating in the region of stable lubrication.

IV. APPLICATION TO DESIGN

It will be noted that equation 7 involves both the generalized operating variable ZN/P, upon which depends the relative film thickness, and the rise in temperature ΔT which determines the operating temperature of the bearing. This suggests the possibility of obtaining a basis for design as discussed previously in this paper by substituting in equation 7 a minimum limiting value of ZN/P to assure stable lubrication, and a maximum limiting value of ΔT based upon a maximum allowable bearing temperature and a maximum temperature of the surroundings. The limiting value for ZN/P could be chosen with a reasonable margin of safety above the minimum point of f-(ZN/P) curves where friction data for similar bearings are available, or it may be based upon operating experience with bearings of the same type. The limiting operating temperature would depend upon the particular application. In some cases the question of oil stability may be the deciding factor, while in others the effect of temperature upon the characteristics of the bearing metal may be of more importance. If such limiting values are substituted it is seen that for a given bearing equation 7 will assume the form

$$PN=K, \tag{8}$$

wherein K represents the product of the maximum allowable pressures and speeds at which the bearing will operate under the conditions prescribed for safety. It should be noted that this equation is not dependent upon the particular form of equations 4 and 6 but is applicable to all types of journal bearings where the heat-dissipation characteristics are independent of the speed of the journal. In some bearing applications the heat-dissipation coefficient may be dependent upon the speed of rotation. In these cases the form of equation 8, more nearly representative of actual conditions, might be $PN^c=K$, where c is less than 1.

Equation 8 may be written also as

$$PV=K' \tag{9}$$

This is a more general relation where a given value of K' is applicable to all bearings having the same C/D ratios, L/D ratios and heat-dissipation characteristics. It is of interest that equation 9 is similar in form to a relation that has been used to a considerable extent in the past as a basis for bearing design. It should be noted, however, that equation 9 is not a "blanket" relationship that is applicable to all conditions. It involves a definite relation between load, speed, viscosity of the oil, and operating temperature, as prescribed by the limits assumed for safe operation.

V. NUMERICAL EXAMPLE

To illustrate the application of equation 8, a numerical example is given in which the following assumptions are made: A 360° bearing with $D=3$ in., $L=3$ in., $C=0.004$ in., conditions for heat dissipation are such that $k_3k_4=0.00116$, minimum allowable $ZN/P=30$, maximum temperature of surroundings$=100°$ F, and maximum allowable

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bearing temperature=250° F. Under these assumptions, $\Delta T=150$, $D/C=750$, and $L/D=1$, hence $\Delta f=0.0018$ (from fig. 1). By substituting the above values in equation 7 and solving for PN a value of 850,000 is obtained, which is the value of K in equation 8. Thus with this bearing under the conditions assumed to represent the limits for safe operation the product of the pressure by the speed should not exceed 850,000, or if expressed as in equation 9, PV should not exceed 667,000.

Values for the individual variables P and N, when using different lubricants, may be obtained by the simultaneous solution of the equations $PN=850,000$ and $ZN/P=30$ for a number of values of Z.

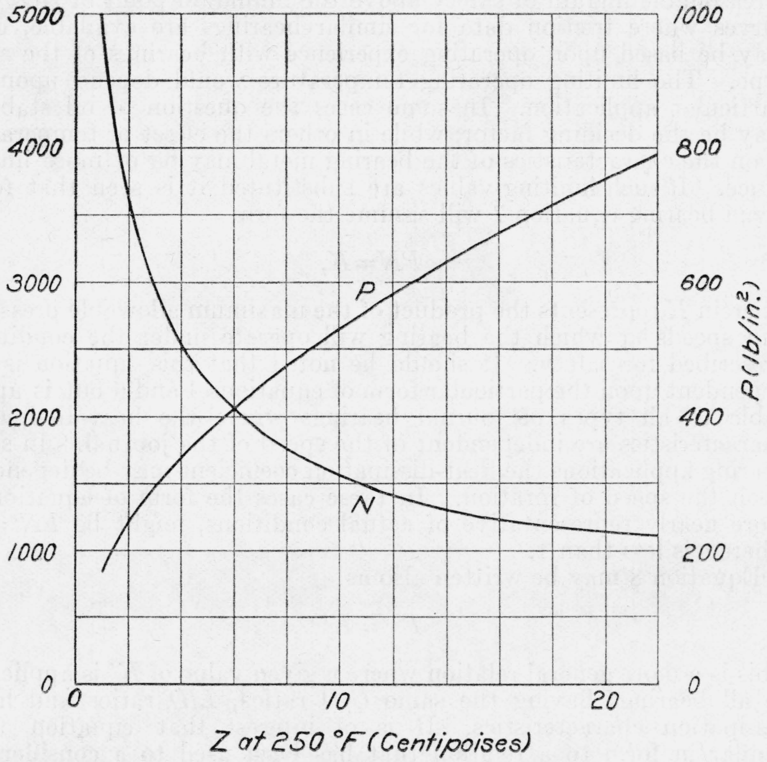

FIGURE 2.—*Pressure-viscosity and speed-viscosity curves for 3- by 3-inch bearing, under assumed limiting conditions.*

The results of these computations are given by the PZ and NZ curves in figure 2. A point on the PZ curve represents the maximum pressure at which the bearing may be operated under the given limitations when using an oil having a viscosity at 250° F as indicated. The point on the NZ curve at the same viscosity indicates the speed required when operating at the maximum pressure. Suppose, for example, an oil having a viscosity of 5 centipoises at 250° F is used in this bearing. From figure 2 the values of P and N at $Z=5$ are 374 lb/in.^{2} and 2,260 rpm, respectively. Thus, if the bearing were operating with this oil at a speed of 2,260 rpm and a load such that the pressure on the projected area was 374 lb/in.^{2}, it would come to equilibrium under the assumed conditions when the temperature reached 250° F and the value of ZN/P would equal 30. It is obvious that if N_1 and P_1 are a pair of limiting values of N and P, any value of P lower than P_1 may be combined with N_1 without reducing the value of ZN/P below the permissible limit or increasing the value of ΔT above the permissible limit, so that a bearing which can be operated at N_1P_1 can be operated safely at N_1 and any value of P below P_1. With a value of P lower than P_1 for a given oil the bearing will not be operating at

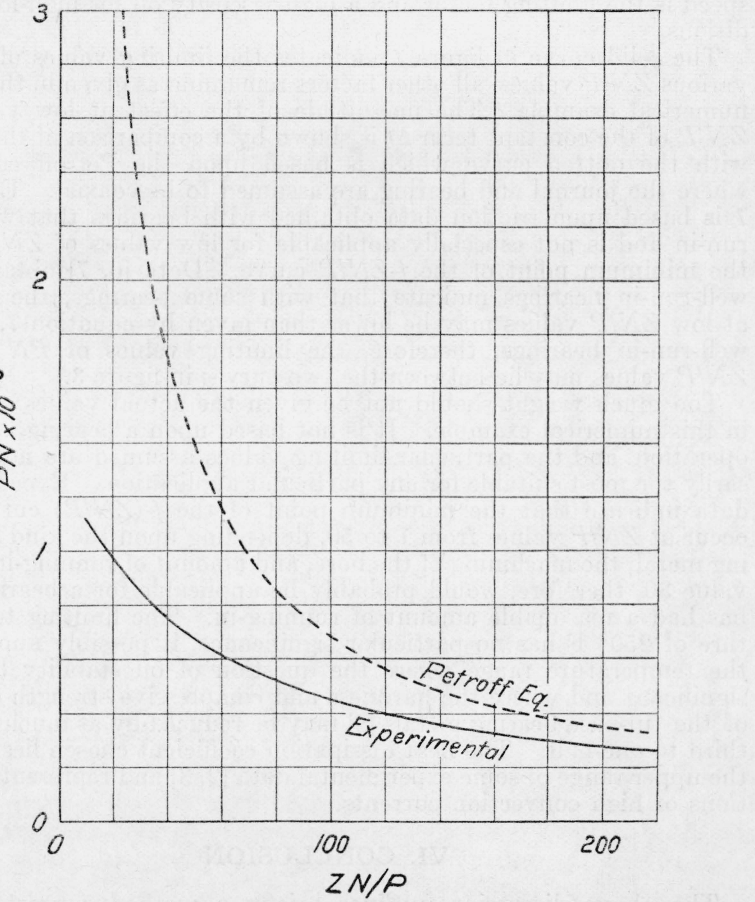

FIGURE 3.—*Curves showing effect of ZN/P value on PN relation for 3- by 3-inch bearing, under assumed limiting conditions.*

$ZN/P=30$ when the limiting temperature is reached; thus the value for PN of 850,000 no longer holds and some value of N other than N_1 becomes the limiting value. This new limiting value may be obtained by the substitution in equation 7 of the limiting value of ΔT, the value of Z at the limiting temperature and the given value of P. Equation 7 will then take on the form $aN\ [bN+\Delta f]=K''$, where a, b, Δf, and K'' are constant terms, and thus may be solved directly for N. Conversely, if the pressure is held at P_1 and N is less than

N_1, the value of PN will be lower than the limit. However, the question as to whether the value of ZN/P is within the limit depends upon whether the lower operating temperature will cause an increase in Z great enough to counteract the decrease in N and hence for a given bearing is dependent upon the viscosity-temperature relation of the particular oil used. The curves in figure 2 also provide a means for selecting a lubricant that will tend to provide maximum bearing capacity for the particular operating conditions. They indicate, for example, the desirability of the use of a low-viscosity oil where high speed is the limiting factor and a high-viscosity oil for high-load conditions.

The solid curve in figure 3 indicates the limiting values of PN at various ZN/P values, all other factors remaining as given in the above numerical example. The magnitude of the effect at low values of ZN/P of the constant term Δf is shown by a comparison of this curve with the dotted curve which is based upon the Petroff equation, where the journal and bearing are assumed to be coaxial. Equation 7 is based upon friction data obtained with bearings that were not run-in and is not especially applicable for low values of ZN/P near the minimum point of the f-ZN/P curve. Data [6, 7] obtained on well-run-in bearings indicate that with some bearings, the friction at low ZN/P values may be lower than given by equation 7. With well-run-in bearings, therefore, the limiting values of PN at low ZN/P values may lie between the two curves in figure 3.

Too much weight should not be given the actual values obtained in this numerical example. It is not based upon a bearing in actual operation, and the particular limiting values assumed are not necessarily the most suitable for any particular application. Experimental data indicate that the minimum point of the f-(ZN/P) curve may occur at ZN/P values from 1 to 50, depending upon the kind of bearing metal, the machining of the bore, and amount of running-in. The value 30, therefore, would probably be applicable for a bearing that has had a reasonable amount of running-in. The limiting temperature of 250° F has no particular significance, it possibly approaches the temperature range where the question of oil stability becomes significant and where the hardness and compressive strength of some of the tin-base bearing metals [8] may be reduced by as much as one-third to one-half. The heat dissipation coefficient chosen lies toward the upper range of some experimental data [2, 3] and represents conditions of high convection currents.

VI. CONCLUSION

The above discussion outlines briefly a method suggested as a possible basis for design more especially for heavy-duty, high-speed bearings. As outlined, the method is directly applicable to self-cooled bearings. It may be made applicable to independently cooled bearings, however, either by the insertion in equation 7 of suitable factors relating to the characteristics of the cooling system, or in its present form to indicate a limit beyond which independent cooling is necessary, and thus provide a measure of the capacity of the cooling system required. It is realized that with the data available at present it will provide only a rough approximation of bearing capacity. Further steps indicated before it could be applied more rigorously are a complete investigation of the frictional characteristics of journal bearings at the lower values of ZN/P and a comprehensive study of the factors affecting the heat-dissipation characteristics for various types of bearing applications.

VII. SYMBOLS

D=journal diameter, in.
L=bearing length, in.
C=running clearance (difference between bearing diameter and journal diameter), in.
W=total load acting on bearing, lb.
N=speed of journal, rpm.
$V=\pi DN/12$=peripheral speed of journal, fpm.
Z=absolute viscosity of lubricant at atmospheric pressure and bearing temperature, centipoises.
F=tangential frictional force, lb.
$f=F/W$=coefficient of friction.
$P=W/LD$=pressure on projected area of bearing, lb/in.^{2}.
H=heat generated in bearing per unit time, Btu/min.
H'=heat dissipated by bearing per unit time, Btu/min.
A=bearing area effective for heat dissipation, in.^{2}.
T_B=bearing temperature, °F.
T_0=temperature of surrounding atmosphere, °F.
$\Delta T=T_B-T_0$=temperature rise.
k_1=mechanical equivalent of heat$=1/778\ (12)$=Btu/in. lb.
Δf=correction factor, function of length-diameter ratio.
k_2=coefficient in equation $3=473\times 10^{-10}$.
k_3=overall coefficient of heat transfer.
$k_4=A/\pi DL$=ratio area effective for heat dissipation to working area of bearing.

VIII. REFERENCES

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